DESCRIPTION

HERMETICALLY SEALED COMPRESSOR

Technical Field

The present invention relates to a hermetically sealed compressor of a kind used in the freezer cycle such as, for example, a combination refrigerator and freezer.

Background Art

With respect to the hermetically sealed compressor of a kind used in a combination refrigerator and freezer for household use, demands have arisen in recent years for sophistication of the compressor to minimize the electric power consumption and to reduce noises. In the face of those demands, attempts have been made and are now in progress to employ a lubricant oil of low viscosity in the compressor and/or to use an inverter-controlled drive system to allow the compressor to be operated under a low speed operating condition (for example, at about 1,200 rpm in the case of the household refrigerator). The hermetically sealed compressor requires a freezer oil to be sufficiently supplied to bearings and sliding elements such as, for example, a connecting rod and a piston, and the use of an oil pump capable of accomplishing a stabilized oil supply even during the low speed operation in order to be effective as an elemental technology.

This kind of oil pump hitherto used is of a type designed for collecting the oil at a position of the large radius of gyration of a rotating element, at which a strong centrifugal force can be obtained. See, for example, Japanese Patent Publication No. H09-512315.

The prior art hermetically sealed compressor of the kind referred to above will now be discussed with reference to the accompanying drawings.

Fig. 15 shows a longitudinal sectional illustration of the prior art

hermetically sealed compressor and Fig. 16 shows an enlarged illustration of a portion of the prior art hermetically sealed compressor. Referring to these figures, the prior art compressor includes a hermetically sealed vessel 1 filled with a quantity of coolant 2 and also with a freezer oil 3.

An electromotive element 11 is made up of a stator 12 electrically connected with an external electric power source (not shown) and a rotor 13 positioned inside the stator 12 with a gap delimited between it and the stator 12.

A compressing element 21 includes a shaft 22, having a main shaft portion 22a with the rotor 13 mounted thereon, and an eccentric shaft portion 22b, a cylinder block 23 fixed above the stator 12 and having a compressing chamber 23a defined therein, a bearing 24 provided in the cylinder block 23 and supporting the main shaft portion 22a, a reciprocating piston 25 reciprocatingly movable within the compressing chamber 23a, and a connecting means 26 for connecting the piston 25 with the eccentric shaft portion 22b, thereby forming a reciprocating compressor mechanism.

The details of an oil supply mechanism will be discussed hereinafter.

The main shaft portion 22a of the shaft 22 has a lower end formed with an oil pump 31 immersed into the freezer oil 3.

The rotor 13 is machined to have an oil delivery hole 32, which provides a fluid connection path in an inner wall thereof, mounted on the main shaft portion 22a, and is positioned at a large radius of gyration on a side adjacent an upper face of the rotor 13 so that the oil delivery hole 32 can provide a fluid connection path between an upper end of the oil pump 31 and a communicating hole 33.

An oil delivery tube 34 is fixedly inserted into an opening of the oil delivery hole 32 adjacent the upper face of the rotor 13, and there is provided an oil collecting means 35 fitted to the bearing 24 for receiving the freezer oil 3, discharged from the oil delivery tube 34, and a supply means 36 for supplying a

freezer oil 3, collected in the oil collecting means 35, to a sliding area made up of the main shaft portion 22a and the bearing 24.

The operation of the hermetically sealed compressor so constructed as hereinabove described will now be described.

When an electric power is supplied to the stator 12 from an external electric power source, the rotor 13 rotates together with the shaft 22. As a result thereof, the eccentric motion of the eccentric shaft portion 22b causes the piston 25 to undergo a reciprocating motion within the compressing chamber 23a through the connecting means 26 to thereby perform a predetermined compressing operation to compress gases being sucked.

The main shaft portion 22a rotates with the rotation of the shaft 22, and the freezer oil 3 flowing upwardly through the oil pump 31 passes through the communicating hole 33 and then flows upwardly through the oil delivery hole 32 and the oil delivery tube 34 by the effect of a centrifugal force.

The freezer oil 3 discharged from the oil delivery tube 34 is poured into the oil collecting means 35.

The freezer oil 3 so collected in the oil collecting means 35 lubricates the sliding area, made up of the main shaft portion 22a and the bearing 24, through the supply means 36.

Also, for enhancing the efficiency for reducing the electric power consumption and for minimizing vibrations and noises, the electric compressor having an electromotive element is employed in the form of an electric motor of a bipolar permanent magnet type having a permanent magnet built in the rotor (such as disclosed in, for example, Japanese Laid-open Patent Publication No. 2001-73948), in place of the induction motor, and the hermetically sealed compressor, in which a hermetically sealed compressor has mechanical units positioned in an upper region (such as disclosed in, for example, Japanese Laid-open Patent Publication No. 2000-110723), are currently available in the art.

Hereinafter this prior art hermetically sealed compressor will be discussed with reference to the accompanying drawings.

Fig. 17 illustrate a longitudinal sectional view of the prior art hermetically sealed compressor disclosed in Japanese Laid-open Patent Publication No. No. 2001-73948 referred to above. As shown in Fig. 17, an electromotive element 54, including a stator 52 and a rotor 53, and a compressing element 55 adapted to be driven by the electromotive element 54 are accommodated within a hermetically sealed vessel 51, and a quantity of lubricant oil 56 is filled within the sealed vessel 51. A shaft 60 has a main shaft portion 61, on which the rotor 53 is fixed, and an eccentric shaft portion 62 formed eccentrically relative to the main shaft portion 61. A cylinder block 64 has a generally tubular compressing chamber 65 and also has a main bearing 67 fixed therein and made of an aluminum material that is a non-magnetic material. A piston 69 is inserted into the compressing chamber 65 in the cylinder block 64 for reciprocating movement therein and is drivingly connected with the eccentric shaft portion 62 by means of a connecting means 70.

An electromotive element 54 is in the form of an electric motor of a bipolar permanent magnet type made up of the stator 52 including a stator iron core 75, prepared from a laminated electromagnetic steel plate having a winding wound therearound, and the rotor 53 having a permanent magnet 77 built therein. Also, a protective end plate 78 for preventing the permanent magnet 77 from dropping out is fixed to the rotor iron core 76.

A hollow bore 79 is provided in one end of the rotor iron core 76 adjacent the compressing element 55, and the main bearing 67 extends inside this hollow bore 79.

With respect to the hermetically sealed compressor so constructed as hereinabove described, the operation thereof will now be described.

The rotor 53 of the electromotive element 54 drives the shaft 60 and

the transmission of the rotary motion of the eccentric shaft portion 62 to the piston 69 through the connecting means 70 results in the piston 69 undergoing a reciprocating motion within the compressing chamber 65. Accordingly, the coolant gas is, after having been sucked from a cooling system (not shown) into and then compressed within the compressing chamber 65, discharged again into the cooling system.

In the description that follows, the flow of magnetic fluxes and the loss thereof will be discussed. Since the main bearing 67 is made of a non-magnetic material, no magnetic force of attraction develops between an inner peripheral surface of the bore 67 and the main bearing 67 and no loss of torque occurs and, also, magnetic fluxes emanating from the permanent magnet 77 will not be attracted by the main bearing 67, since the main bearing 67 is made of the non-magnetic material, but most of them pass only through the rotor iron core 76. Accordingly, no iron loss (particularly, the eddy current loss) occurs virtually within the main bearing 67 and, thus, a high efficiency can be obtained.

Fig. 18 illustrates a longitudinal sectional view of the prior art hermetically sealed compressor disclosed in Japanese Laid-open Patent Publication No. No. 2000-110723 referred to above.

As shown in Fig. 18, an electromotive element 84, made up of a stator 82 and a rotor 83, and a compressing element 85 adapted to be driven by the electromotive element 84 are accommodated within a hermetically sealed vessel 81, and a lubricant oil 86 is reserved within the hermetically sealed vessel 81. A shaft 87 has a main shaft portion 98, on which the rotor 83 is press-fitted, and an eccentric shaft portion 89 eccentrically formed relative to the main shaft portion 88.

A cylinder block 94 has a generally cylindrical compressing chamber defined therein and also has a main bearing 96 for rotatably supporting the main shaft portion 88. A piston 87 is inserted within the compressing chamber in the cylinder block 94 for reciprocating movement therein and is drivingly connected with

the eccentric shaft portion 89 through a connecting means 98.

The shaft 87 has oil feed passages 90 and 91 defined therein, and the main shaft portion 88 has an outer periphery formed with a spiral groove 92 defined therein so as to extend spirally upwardly in a direction counter to the direction of rotation of the shaft 87, which groove 92 has a lower end communicated with a portion proximate to an upper end of the oil feed passage 90. An upper end of the spiral groove 92 is communicated with a portion proximate to the oil feed passage 91. An oil cone 93 having one end opening into a quantity of lubricant oil 86 and the opposite end communicated with the oil feed passage 90 is fixed on a lower end of the main shaft portion 88. A unit including the compressing element 85 and the electromotive element 84 is elastically supported within the hermetically sealed vessel 81 by means of a spring 95 arranged below the electromotive element 84.

With respect to the hermetically sealed compressor so constructed as hereinabove described, the operation thereof will now be described.

The rotor 83 of the electromotive element 84 drives the shaft 87 and the transmission of the rotary motion of the eccentric shaft portion 99 to the piston 97 through the connecting means 98 results in the piston 97 undergoing a reciprocating motion within the compressing chamber. Accordingly, the coolant gas is, after having been sucked from a cooling system (not shown) into and then compressed within the compressing chamber, discharged again into the cooling system.

On the other hand, the oil cone 93 performs a pumping function when the shaft 87 is driven. By the effect of the pumping function of the oil cone 93, the lubricant oil 86 in a bottom region of the hermetically sealed vessel 81 is drawn upwardly through the oil feed passage 90. The lubricant oil 96 so drawn upwardly to an upper area of the oil feed passage 90 is introduced into the spiral groove 92. Since the spiral groove 92 is inclined in the same direction as an inertia force acting in a direction counter to the direction of rotation of the shaft 87, a large force of

transportation acts on the lubricant oil 86 to further draw the latter upwardly.

The lubricant oil 86 is not only drawn upwardly through the spiral groove 92, but also supplied towards a sliding area of the shaft 87. The lubricant oil 86 arriving at the upper end of the spiral groove 92 flows into the oil feed passage 91 and is then supplied to and lubricates a sliding area such as the eccentric shaft portion 89 and others.

Also, although considerable vibrations of the compressing element 85 in the vicinity of the piston 97 would occur since the piston 97 undergoes the reciprocating movement, the vibration occurring below the electromotive element 84 distant from the piston 97 will be low. Since the unit of the compressing element 85 and the electromotive element 84 is elastically supported by the spring 95 at a region below the electromotive element 84, where the vibration is low, the vibration transmitted to the hermetically sealed vessel 81 through the spring 95 can be suppressed, allowing the hermetically sealed compressor not to produce considerable vibrations.

It has, however, been found that the construction disclosed in Japanese Patent Publication No. H09-512315 referred to above has a problem in that since the complicated oil feed passages are formed in the rotor, the cost of manufacture of the hermetically sealed compressor tends to become high. Also, since the oil delivery tube, the oil collecting means and the supply means are required for the transfer of the freezer oil from the position of the large radius of gyration to the position of the small radius of gyration against the centrifugal force, there is a risk of the supply of the lubricant oil in the hermetically sealed compressor being unstable.

If the electric motor of the bipolar permanent magnet type utilized in Japanese Laid-open Patent Publication No. 2001-73948 for enhancing the efficiency is employed in the relatively silent hermetically sealed compressor constructed according to Japanese Laid-open Patent Publication No. 2000-110723 referred to

above, formation of the oil feed passage 90 having a large cross-sectional area in the main shaft portion 88 will result in the presence of a space inside the rotor iron core 76, where no magnetic circuit can be formed, but only a partially narrow magnetic circuit can be formed accompanied by an increase of the magnetic resistance and, therefore, a problem has been found in that the amount of magnetic fluxes produced by the permanent magnet 77 and emanating from the rotor iron core 76 tends to be small as compared with that produced in the absence of the oil feed passage 90, resulting in increase of the loss.

SUMMARY OF THE INVENTION

The present invention has been developed to overcome the above-described disadvantages.

It is accordingly an objective of the present invention to provide a highly reliable, inexpensive hermetically sealed compressor, in which a stable oil supply is possible during a low speed operating condition.

Another objective of the present invention is to provide a low vibration, highly efficient hermetically sealed compressor, which when the electric motor of the bipolar permanent magnet type is applied in the hermetically sealed compressor of the type having the compressing element arranged in an upper region, the amount of magnetic fluxes produced by the permanent magnet can be increased to enhance the efficiency.

Disclosure of the Invention

In accomplishing the above and other objectives, the present invention in accordance with a preferred embodiment thereof provides a hermetically sealed compressor which includes a sealed vessel filled with a coolant and a freezer oil, an electromotive element including a rotor and a stator accommodated within the sealed vessel, and a compressing element accommodated within an upper region of the sealed vessel and adapted to be driven by the electromotive element. This compressing element is provided with a shaft, arranged so as to extend vertically

and having the rotor mounted thereon, and a bearing for supporting the shaft. The hermetically sealed compressor also includes a first oil pump provided in a lower portion of the shaft and opening into the freezer oil, a second oil pump provided above the first oil pump and formed by a spiral groove, provided on an outer periphery of the shaft, and an inner peripheral wall surface of the rotor, and a third oil pump provided above the second oil pump and formed by a spiral groove, provided on the outer periphery of the shaft, and an inner peripheral surface of the bearing.

According to the present invention, the supply of the freezer oil can be accomplished stably even at the low speed operating condition by allowing the freezer oil to ascend through the first oil pump by the effect of a centrifugal force under the pumping head secured by the second oil pump. Also, with a minimized number of component parts used, the compressor can be constructed more easily and, therefore, it is possible to provide a highly reliable, inexpensive hermetically sealed compressor.

If the spiral groove of the second oil pump and the spiral groove of the third oil pump are formed continuously, formation of a spiral groove on the outer periphery of the shaft can be carried out continuously, resulting in enhancement of the mass production.

Also, if the spiral groove of the second oil pump and the spiral groove of the third oil pump are so designed as to open in communication with a first gap formed between the rotor and the bearing, no sliding will occur between the rotor and the bearing and, therefore, reduction in noise and electric power consumption can be attained.

When the first gap is chosen to be 0.5 mm or smaller over the entire circumference thereof, the freezer oil flowing outwardly from the first gap can be minimized and, on the other hand, the amount of the oil to be supplied to the sliding area can be increased, resulting in increase of the reliability.

Formation of the bore in the upper end face of the rotor for receiving the bearing and, also, formation of the second gap between an inner peripheral surface of the bore and an outer peripheral surface of the bearing allows the freezer oil, pooled within the second gap, to provide a resistance, with which flow of the freezer oil outwardly from the second gap can be suppressed.

If the second gap is provided with a site of 1.0 mm or smaller over the entire circumference thereof, the freezer oil flowing outwardly through the second gap can be minimized by the effect of a resistance brought about by the viscosity of the freezer oil, allowing an increased amount of the oil to be supplied to the sliding area.

If the depth of the bore is set to be 5.0 mm or larger, it is possible to suppress an undesirable flow of the freezer oil pooled within the second gap outwardly from the second gap.

Also, when the axially elastically deformable washer is interposed in the first gap, the first gap will become almost zero over the entire circumference thereof and, accordingly, the freezer oil tending to flow outwardly from the first gap can be drastically reduced to allow it to be supplied to the sliding area in an increased quantity.

Where the center of magnetism of the rotor is displaced below a center of magnetism of the stator, the first gap becomes almost zero over the entire circumference thereof as the rotor ascends by the effect of the magnetic force of attraction during an operation and, therefore, the freezer oil tending to flow outwardly from the first gap can be drastically reduced to allow it to be supplied to the sliding area in an increased quantity.

The present invention in accordance with another preferred embodiment also provides a hermetically sealed compressor which includes a sealed vessel filled with a lubricant oil, an electromotive element including a rotor and a stator and accommodated within the sealed vessel, and a compressing

element accommodated within the sealed vessel and adapted to be driven by the electromotive element. This compressing element is provided with a shaft, having an eccentric shaft portion and a main shaft portion, and a main bearing for supporting the shaft. In this compressor, a first oil pump is provided in a lower portion of the shaft and opening into the lubricant oil, a second oil pump is provided above the first oil pump and formed by a spiral groove, provided on an outer periphery of the shaft, and an inner peripheral wall surface of the rotor, and a third oil pump is provided above the second oil pump and formed by a spiral groove, provided on the outer periphery of the shaft, and an inner peripheral surface of the main bearing. The electromotive element referred to above is employed in the form of a bipolar permanent magnet electric motor including a permanent magnet built in a rotor iron core of the rotor.

According to this aspect of the present invention, no large space such as the oil feed passage hitherto employed in the prior art compressor exists inside the main shaft portion and, hence, there is nothing that may interfere with the magnetic path inside the rotor iron core. Therefore, a relatively large magnetic path can be formed and the amount of the magnetic fluxes produced inside the rotor iron core can increase, resulting in reduction of the loss to thereby increase the efficiency of the electromotive element.

If the main bearing is so configured that it does not intersect a plane containing one end of the rotor iron core adjacent the compressing element and lying generally perpendicular to a longitudinal axis of the main shaft, the use of the bore hitherto employed for receiving the main shaft portion in the rotor iron core can be eliminated and, hence, narrowing of the magnetic path, which would result from the use of such bore, can be eliminated, resulting in an increase of the amount of the magnetic flux inside the rotor iron core to thereby increase the efficiency further.

In the even that an auxiliary shaft portion is provided coaxially of the main shaft portion with the eccentric shaft portion intervening between it and the

main shaft portion, together with an auxiliary bearing for supporting the auxiliary shaft portion, the auxiliary bearing will act to essentially regulate the inclination of the shaft and, therefore, even though the main bearing may have a reduced length and is not insertable into the rotor iron core, the inclination of the shaft will not vary virtually and neither the shaft nor the main bearing and the auxiliary bearing will become complicated, resulting in an increase of the reliability and the efficiency while the noise is suppressed.

Also, the bipolar permanent magnet electric motor is employed in the form of a self-starting permanent magnet synchronous motor including a rotor having a plurality of conductor bars of a starter cage conductor in an outer periphery of the rotor iron core and also having a plurality of permanent magnets embedded within the rotor iron core, a synchronous motor effective to provide high efficiency can be employed in the compressor of the present invention.

Preferably, each of the permanent magnets is in the form of a rare earth magnet, so that a strong magnetic force can be obtained, allowing not only the electric motor to be manufactured in a compact size and lightweight, but also the hermetically sealed compressor to be manufactured in a compact size and lightweight.

Brief Description of the Drawings

The above and other objectives and features of the present invention will become more apparent from the following description of preferred embodiments thereof with reference to the accompanying drawings, throughout which like parts are designated by like reference numerals, and wherein:

Fig. 1 is a longitudinal sectional view of a hermetically sealed compressor according to a first preferred embodiment of the present invention;

Fig. 2 is an enlarged view showing an important portion of the hermetically sealed compressor according to the first embodiment;

Fig. 3 is a longitudinal sectional view of the hermetically sealed

compressor according to a second preferred embodiment of the present invention;

Fig. 4 is an enlarged view showing an important portion of the hermetically sealed compressor according to the second embodiment;

Fig. 5 is a longitudinal sectional view of the hermetically sealed compressor according to a third preferred embodiment of the present invention;

Fig. 6 is an enlarged view showing an important portion of the hermetically sealed compressor according to the third embodiment;

Fig. 7 is a longitudinal sectional view of the hermetically sealed compressor according to a fourth preferred embodiment of the present invention in a halted condition;

Fig. 8 is an enlarged view showing an important portion of the hermetically sealed compressor according to the fourth embodiment in a halted condition;

Fig. 9 is a longitudinal sectional view of the hermetically sealed compressor according to the fourth embodiment in an operated condition;

Fig. 10 is a longitudinal sectional view of the hermetically sealed compressor according to a fifth preferred embodiment of the present invention;

Fig. 11 is a sectional view of a rotor employed in the hermetically sealed compressor according to the fifth embodiment;

Fig. 12 is the sectional view of the rotor when an oil feed passage is formed in a main shaft portion;

Fig. 13 is a longitudinal sectional view of the hermetically sealed compressor according to a sixth preferred embodiment of the present invention;

Fig. 14 is a sectional view of the rotor employed in the hermetically sealed compressor according to the sixth embodiment;

Fig. 15 is a longitudinal sectional view of a prior art hermetically sealed compressor;

Fig. 16 is an enlarged view showing an important portion of the prior

art hermetically sealed compressor shown in Fig. 15;

Fig. 17 is a longitudinal sectional view of another prior art hermetically sealed compressor; and

Fig. 18 is a longitudinal sectional view of a further prior art hermetically sealed compressor.

Detailed Description of the Preferred Embodiments

Preferred embodiments of the present invention will be hereinafter described in detail with reference to the accompanying drawings.

Embodiment 1

Fig. 1 is a longitudinal sectional view of a hermetically sealed compressor according to a first preferred embodiment of the present invention and Fig. 2 is an enlarged view showing an important portion of the hermetically sealed compressor according to the first embodiment.

Referring to Figs. 1 and 2, a hermetically sealed vessel 101 is filled with a coolant 102 and reserves a quantity of freezer oil 103. The coolant 102 is R600a, which is a hydrocarbon coolant, whereas the freezer oil 103 is an oil compatible with the coolant 102 such as, for example, synthetic oil, mineral oil or polyol ester oil.

An electromotive element 111 includes a stator 112, electrically connected with an external electric power source (not shown), and a rotor 113 disposed with a predetermined gap intervening between it and an inner side of the stator 112.

A compressing element 121 includes a shaft 122 having a main shaft portion 122a, on which the rotor 113 is mounted, and an eccentric shaft portion 112b, a cylinder block 123 fixed above the stator 112 and defining a compressing chamber 123a therein, a bearing 124 provided in the cylinder block 123 and rotatably supporting the main shaft portion 122a, a piston 125 capable of undergoing a reciprocating motion within the compressing chamber 123a, and a

connecting means 126 for connecting the piston 125 with the eccentric shaft portion 122b, thereby forming a reciprocating compressor mechanism.

A first gap 131 is formed between a lower end face of the bearing 124 and an upper end face of the rotor 113. This first gap 131 is so formed as to have a size of 0.5 mm or smaller over the entire circumference.

The structure of an oil supply mechanism will now be described in detail.

A first oil pump 141 is made up of an inclined hole 142 inclined upwardly within the main shaft portion 122a, immersed into the freezer oil 103, from a lower end face of the main shaft portion 122a, an agitating plate 143 press-fitted into the inclined hole 142, and an end plate 144 having a throughhole 144a extending in a center area thereof and engaged to an opening of the inclined hole 142, to thereby form a centrifugal pump.

A second oil pump 151 is provided above the first oil pump 141 and is made up of a spiral groove 152, provided in an outer periphery of the main shaft portion 122a, and a radially inner wall surface of the rotor 113, to thereby form an inertia pump. The first oil pump 141 and the second oil pump 151 are communicated with each other through the throughhole 153.

A third oil pump 161 is provided above the second oil pump 151 and is made up of a spiral groove 162, which is formed in continuance with the spiral groove 152 formed in the outer periphery of the main shaft portion 122a, and an inner peripheral surface of the bearing 124, to thereby form a viscous pump.

The spiral groove 152 and the spiral groove 162 are formed as a continuous groove traversing the first gap 131 in the form of a helical groove of the same angle of pitch.

The operation and effects of the hermetically sealed compressor so constructed as hereinbefore will now be described.

When electric power is supplied from the external electric power

source to the stator 112, the rotor 113 rotates together with the shaft 122. The eccentric motion of the eccentric shaft portion 122b taking place during the rotation of the shaft 122 causes the piston 125 to undergo a reciprocating motion within the compressing chamber 123a through the connecting means 126 to perform a predetermined compressing function to compress the gas being sucked.

In the following description, the operation to supply the oil will be discussed.

In the first oil pump 141, as the main shaft portion 122a rotates, the freezer oil 103 is swirled within the inclined hole 142 by the agitating plate 143 immersed in the freezer oil 103, and the freezer oil 103 ascends along an inner peripheral wall surface of the inclined hole 142 by the action of a centrifugal force developed as a result of the swirling motion of the freezer oil 103.

The position of the throughhole 153 may be within a range of the main shaft portion 122a on which the rotor 113 is mounted, and the oil supply performance can be increased if the inclined hole 142 of the first oil pump 141, through which the oil is supplied by the utilization of the centrifugal force, is so designed as to have a large diameter. Also, the first oil pump 141 may be considered sufficient if it can provide a pumping head enough to pump the oil up to the second oil pump 151 and, accordingly, even at a low rotational speed of, for example, 20 Hz, the freezer oil 103 can assuredly reach the throughhole 153.

The freezer oil 103 introduced from the first oil pump 141 into the second oil pump 151 through the throughhole 153 ascends within the spiral groove 152 in the second oil pump 151 by the effect of an inertia force developed by the inclination in the spiral groove 152 so as to act in an upward direction.

The freezer oil 103 having moved past the first gap 131 and arriving at the third oil pump 161 ascends within the spiral groove 162 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 124 and the rotating main shaft portion 122a.

The third oil pump 161 as hereinabove described forms a viscous pump that utilizes the viscous force developed by the difference in relative rotation between the bearing 124 and the rotating main shaft portion 122a. In general, since the viscous pump utilizes the viscous force to develop a powerful force of transportation, the freezer oil 103 can be assuredly pushed upwardly even at a low rotational speed of, for example, 20 Hz.

Hence, the freezer oil 103 reaching the third oil pump 161 is utilized to lubricate a sliding surface formed by an outer peripheral surface of the main shaft portion 122a and an inner peripheral surface of the bearing 124 and is also supplied towards the eccentric shaft portion 122b.

According to the foregoing embodiment, the freezer oil 103 can be supplied to various sliding areas assuredly even at the low speed operation and, hence, the highly reliable hermetically sealed compressor could have been realized.

In the foregoing embodiment, what is newly required to form the oil pump is only the agitating plate 143 and the end plate 144, both of which can be inexpensively prepared from an iron plate by the use of a press work. Also, since the spiral groove 152 forming the second oil pump 151 and the spiral groove 162 forming the third oil pump 161 are in the form of a helical groove of the same angle of pitch, machining to form the spiral groove can be applied to the outer periphery of the main shaft portion 122a continuously at a constant speed, facilitating the mass production thereof.

Also, since the first gap 131 is provided between the lower end face of the bearing 124 and the upper end face of the rotor 113, no sliding occurs between the bearing 124 and the rotor 113 and, therefore, neither sliding sound nor the sliding loss occurs, resulting in reduction in noise and also in electric power consumption.

As the freezer oil 103 flows through the first gap 131, a portion of the freezer oil 103 flows radially outwardly from the first gap 131 by the effect of the

centrifugal force and the hydraulic pressure. However, according to the foregoing embodiment, the first gap 131 has a size of 0.5 mm or smaller over the entire circumference thereof and it has been ascertained that the gap of this size will not bring about a considerable reduction of the amount of oil supplied.

It is to be noted that although the foregoing embodiment has been described, in which the diameter of the inclined hole 142 is increased, similar effect can be equally obtained even though the angle of inclination of the hole 142 is increased.

Embodiment 2

Fig. 3 is a longitudinal sectional view of the hermetically sealed compressor according to a second preferred embodiment of the present invention and Fig. 4 is an enlarged view showing an important portion of the hermetically sealed compressor according to the second embodiment.

It is to be noted that reference numerals similar to those used in connection with the first embodiment are used to designate like parts and therefore the details of those parts are not reiterated.

Referring to Figs. 3 and 4, an electromotive element 180 includes a stator 112, electrically connected with an external electric power source (not shown), and a rotor 171 disposed with a predetermined gap intervening between it and an inner side of the stator 112.

A compressing element 185 includes a shaft 122 having a main shaft portion 122a, on which the rotor 171 is mounted, and an eccentric shaft portion 112b, a cylinder block 183 fixed above the stator 112 and defining a compressing chamber 183a therein, a bearing 184 provided in the cylinder block 183 and rotatably supporting the main shaft portion 122a, a piston 125 capable of undergoing a reciprocating motion within the compressing chamber 183a, and a connecting means 126 for connecting the piston 125 with the eccentric shaft portion 122b, thereby forming a reciprocating compressor mechanism.

The rotor 171 has a bore 172 defined therein, from which the bearing 184 extends outwardly towards one side adjacent an upper face of the rotor 171, and a second gap 173 is formed between an inner peripheral surface of the bore 172 and an outer peripheral surface of the bearing 184.

The structure of an oil supply mechanism will now be described in detail.

The first oil pump 141 is made up of an inclined hole 142 inclined upwardly within the main shaft portion 122a from a lower end face of the main shaft portion 122a that is immersed into the freezer oil 103, an agitating plate 143 press-fitted into the inclined hole 142, and an end plate 144 having a throughhole 144a extending in a center area thereof and engaged to an opening of the inclined hole 142, to thereby form a centrifugal pump.

The second oil pump 151 is provided above the first oil pump 141 and is made up of a spiral groove 152, provided in an outer periphery of the main shaft portion 122a, and a radially inner wall surface of the rotor 171, to thereby form an inertia pump. The first oil pump 141 and the second oil pump 151 are communicated with each other through the throughhole 153.

The third oil pump 161 is provided above the second oil pump 151 and is made up of a spiral groove 162, which is formed in continuance with the spiral groove 152 formed in the outer periphery of the main shaft portion 122a, and an inner peripheral surface of the bearing 184, to thereby form a viscous pump.

The spiral groove 152 and the spiral groove 162 are formed as a continuous groove traversing the first gap 131 in the form of a helical groove of the same angle of pitch.

The operation and effects of the hermetically sealed compressor so constructed as hereinbefore will now be described.

When electric power is supplied from the external electric power source to the stator 112, the rotor 171 rotates together with the shaft 122. The

eccentric motion of the eccentric shaft portion 122b taking place during the rotation of the shaft 122 causes the piston 125 to undergo a reciprocating motion within the compressing chamber 183a through the connecting means 126 to perform a predetermined compressing function to compress the gas being sucked.

In the following description, the operation to supply the oil will be discussed.

In the first oil pump 141, as the main shaft portion 122a rotates, the freezer oil 103 is swirled within the inclined hole 142 by the agitating plate 143 immersed in the freezer oil 103, and the freezer oil 103 ascends along an inner peripheral wall surface of the inclined hole 142 by the action of a centrifugal force developed as a result of the swirling motion of the freezer oil 103.

The position of the throughhole 153 may be within a range of the main shaft portion 122a on which the rotor 171 is mounted, and the oil supply performance can be increased if the inclined hole 142 of the first oil pump 141, through which the oil is supplied by the utilization of the centrifugal force, is so designed as to have a large diameter. Also, the first oil pump 141 may be considered sufficient if it can provide a pumping head enough to pump the oil up to the second oil pump 151 and, accordingly, even at a low rotational speed of, for example, 20 Hz, the freezer oil 103 can assuredly reach the throughhole 153.

The freezer oil 103 introduced from the first oil pump 141 into the second oil pump 151 through the throughhole 153 ascends within the spiral groove 152 in the second oil pump 151 by the effect of an inertia force developed by the inclination in the spiral groove 152 so as to act in an upward direction.

The freezer oil 103 having moved past the first gap 131 and arriving at the third oil pump 161 ascends within the spiral groove 162 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 184 and the rotating main shaft portion 122a.

The third oil pump 161 as hereinabove described forms a viscous

pump that utilizes the viscous force developed by the difference in relative rotation between the bearing 184 and the rotating main shaft portion 122a. In general, since the viscous pump utilizes the viscous force to develop a powerful force of transportation, the freezer oil 103 can be assuredly pushed upwardly even at a low rotational speed of, for example, 20 Hz.

Hence, the freezer oil 103 reaching the third oil pump 161 is utilized to lubricate a sliding surface formed by an outer peripheral surface of the main shaft portion 122a and an inner peripheral surface of the bearing 184 and is also supplied towards the eccentric shaft portion 122b.

According to the foregoing second embodiment, the freezer oil 103 can be supplied to various sliding areas assuredly even at the low speed operation and, hence, the highly reliable hermetically sealed compressor could have been realized.

In the foregoing second embodiment, what is newly required to form the oil pump is only the agitating plate 143 and the end plate 144, both of which can be inexpensively prepared from an iron plate by the use of a press work. Also, since the spiral groove 152 forming the second oil pump 151 and the spiral groove 162 forming the third oil pump 161 are in the form of a helical groove of the same angle of pitch, machining to form the spiral groove can be applied to the outer periphery of the main shaft portion 122a continuously at a constant speed, facilitating the mass production thereof.

Also, since the first gap 131 is provided between the lower end face of the bearing 184 and the upper end face of the rotor 171, no sliding occurs between the bearing 184 and the rotor 171 and, therefore, neither sliding sound nor the sliding loss occurs, resulting in reduction in noise and also in electric power consumption.

As the freezer oil 103 flows through the first gap 131, a portion of the freezer oil 103 flows radially outwardly from the first gap 131 by the effect of the

centrifugal force and the hydraulic pressure. However, in this embodiment, the gravity of the freezer oil 103 reserved within the second gap 173 provides a resistance, with which the freezer oil 103 flowing outwardly from the second gap 173 can be reduced, resulting in increase of the reliability.

Also, although if the second gap 173 formed between the inner peripheral surface of the bore 172 and the outer peripheral surface of the bearing 184 is large, the amount of the freezer oil 103 flowing outwardly from such gap will increase, accompanied by reduction of the amount of the oil supplied, it has been ascertained that the reduction of the amount of the oil supplied can be abruptly minimized if the second gap 173 is so designed as to have a portion of 1.0 mm or smaller over the entire length thereof.

Similarly, if the bore 172 is designed to have a depth of 5.0 mm or greater over the entire length thereof, it has been ascertained that an outward flow of the freezer oil 103 can be avoided by the effect of the gravity of the freezer oil 103 and, therefore, no reduction of the amount of the oil supplied occurs virtually.

It is to be noted that although the foregoing embodiment has been described, in which the diameter of the inclined hole 142 is increased, similar effect can be equally obtained even though the angle of inclination of the hole 142 is increased.

Embodiment 3

Fig. 5 is a longitudinal sectional view of the hermetically sealed compressor according to a third preferred embodiment of the present invention and Fig. 6 is an enlarged view showing an important portion of the hermetically sealed compressor according to the third embodiment;

It is to be noted that reference numerals similar to those used in connection with the first embodiment are used to designate like parts and therefore the details of those parts are not reiterated.

Referring to Figs. 5 and 6, the first oil pump 141 is made up of an

inclined hole 142 inclined upwardly within the main shaft portion 122a from a lower end face of the main shaft portion 122a that is immersed into the freezer oil 103, an agitating plate 143 press-fitted into the inclined hole 142, and an end plate 144 having a throughhole 144a extending in a center area thereof and engaged to an opening of the inclined hole 142, to thereby form a centrifugal pump.

The second oil pump 151 is provided above the first oil pump 141 and is made up of a spiral groove 152, provided in an outer periphery of the main shaft portion 122a, and a radially inner wall surface of the rotor 113, to thereby form an inertia pump. The first oil pump 141 and the second oil pump 151 are communicated with each other through the throughhole 153.

The third oil pump 161 is provided above the second oil pump 151 and is made up of a spiral groove 162, which is formed in continuance with the spiral groove 152 formed in the outer periphery of the main shaft portion 122a, and an inner peripheral surface of the bearing 124, to thereby form a viscous pump.

The spiral groove 152 and the spiral groove 162 are formed as a continuous groove traversing the first gap 131 in the form of a helical groove of the same angle of pitch.

An axially elastically deformable washer 181 is interposed in the first gap 131 defined between the lower end face of the bearing 124 and the upper end face of the rotor 113.

The operation and effects of the hermetically sealed compressor so constructed as hereinbefore will now be described.

When electric power is supplied from the external electric power source to the stator 112, the rotor 113 rotates together with the shaft 122. The eccentric motion of the eccentric shaft portion 122b taking place during the rotation of the shaft 122 causes the piston 125 to undergo a reciprocating motion within the compressing chamber 123a through the connecting means 126 to perform a predetermined compressing function to compress the gas being sucked.

In the following description, the operation to supply the oil will be discussed.

In the first oil pump 141, as the main shaft portion 122a rotates, the freezer oil 103 is swirled within the inclined hole 142 by the agitating plate 143 immersed in the freezer oil 103, and the freezer oil 103 ascends along an inner peripheral wall surface of the inclined hole 142 by the action of a centrifugal force developed as a result of the swirling motion of the freezer oil 103.

The position of the throughhole 153 may be within a range of the main shaft portion 122a on which the rotor 113 is mounted, and the oil supply performance can be increased if the inclined hole 142 of the first oil pump 141, through which the oil is supplied by the utilization of the centrifugal force, is so designed as to have a large diameter. Also, the first oil pump 141 may be considered sufficient if it can provide a pumping head enough to pump the oil up to the second oil pump 151 and, accordingly, even at a low rotational speed of, for example, 20 Hz, the freezer oil 103 can assuredly reach the throughhole 153.

The freezer oil 103 introduced from the first oil pump 141 into the second oil pump 151 through the throughhole 153 ascends within the spiral groove 152 in the second oil pump 151 by the effect of an inertia force developed by the inclination in the spiral groove 152 so as to act in an upward direction.

The freezer oil 103 having moved past the first gap 131 and arriving at the third oil pump 161 ascends within the spiral groove 162 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 184 and the rotating main shaft portion 122a.

The freezer oil 103 reaching the third oil pump 161 after having flown across an inner peripheral wall surface of the washer 181 ascends within the spiral groove 162 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 124 and the rotating main shaft portion 122a.

The third oil pump 161 as hereinabove described forms a viscous

pump that utilizes the viscous force developed by the difference in relative rotation between the bearing 124 and the rotating main shaft portion 122a. In general, since the viscous pump utilizes the viscous force to develop a powerful force of transportation, the freezer oil 103 can be assuredly pushed upwardly even at a low rotational speed of, for example, 20 Hz.

Hence, the freezer oil 103 reaching the third oil pump 161 is utilized to lubricate a sliding surface formed by an outer peripheral surface of the main shaft portion 122a and an inner peripheral surface of the bearing 124 and is also supplied towards the eccentric shaft portion 122b.

Since the axially elastically deformable washer 181 is interposed in the first gap 131 defined between the lower end face of the bearing 124 and the upper end face of the rotor 113, the freezer oil 103 will not virtually flow outwardly from the first gap 131.

Therefore, according to the foregoing third embodiment, the freezer oil 103 can be supplied to various sliding areas assuredly even at the low speed operation and, hence, the highly reliable hermetically sealed compressor could have been realized.

In the foregoing third embodiment, what is newly required to form the oil pump is only the agitating plate 143 and the end plate 144, both of which can be inexpensively prepared from an iron plate by the use of a press work. Also, since the spiral groove 152 forming the second oil pump 151 and the spiral groove 162 forming the third oil pump 161 are in the form of a helical groove of the same angle of pitch, machining to form the spiral groove can be applied to the outer periphery of the main shaft portion 122a continuously at a constant speed, facilitating the mass production thereof.

It is to be noted that although the foregoing embodiment has been described, in which the diameter of the inclined hole 142 is increased, similar effect can be equally obtained even though the angle of inclination of the hole 142 is

increased.

Embodiment 4

Fig. 7 is a longitudinal sectional view of the hermetically sealed compressor according to a fourth preferred embodiment of the present invention in a halted condition, Fig. 8 is an enlarged view showing an important portion of the hermetically sealed compressor according to the fourth embodiment in the halted condition, and Fig. 9 is a longitudinal sectional view of the hermetically sealed compressor according to the fourth embodiment in an operated condition;

It is to be noted that reference numerals similar to those used in connection with the first embodiment are used to designate like parts and therefore the details of those parts are not reiterated.

Referring to Figs. 7 and 8, the rotor 113 is so arranged that the center of the magnetism of the rotor 113 can be displaced downwardly from the center of magnetism of the stator 112. The amount of this displacement is chosen to be larger than the first gap formed between the lower end face of the bearing 124 and the upper end face of the rotor 113.

The operation and effects of the hermetically sealed compressor so constructed as hereinbefore will now be described.

When electric power is supplied from the external electric power source to the stator 112, the rotor 113 rotates together with the shaft 122. The eccentric motion of the eccentric shaft portion 122b taking place during the rotation of the shaft 122 causes the piston 125 to undergo a reciprocating motion within the compressing chamber 123a through the connecting means 126 to perform a predetermined compressing function to compress the gas being sucked.

In the following description, the operation to supply the oil will be discussed.

In the first oil pump 141, as the main shaft portion 122a rotates, the freezer oil 103 is swirled within the inclined hole 142 by the agitating plate 143

immersed in the freezer oil 103, and the freezer oil 103 ascends along an inner peripheral wall surface of the inclined hole 142 by the action of a centrifugal force developed as a result of the swirling motion of the freezer oil 103.

The position of the throughhole 153 may be within a range of the main shaft portion 122a on which the rotor 113 is mounted, and the oil supply performance can be increased if the inclined hole 142 of the first oil pump 141, through which the oil is supplied by the utilization of the centrifugal force, is so designed as to have a large diameter. Also, the first oil pump 141 may be considered sufficient if it can provide a pumping head enough to pump the oil up to the second oil pump 151 and, accordingly, even at a low rotational speed of, for example, 20 Hz, the freezer oil 103 can assuredly reach the throughhole 153.

The freezer oil 103 introduced from the first oil pump 141 into the second oil pump 151 through the throughhole 153 ascends within the spiral groove 152 in the second oil pump 151 by the effect of an inertia force developed by the inclination in the spiral groove 152 so as to act in an upward direction.

The freezer oil 103 having moved past the first gap 131 and arriving at the third oil pump 161 ascends within the spiral groove 162 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 124 and the rotating main shaft portion 122a.

The third oil pump 161 as hereinabove described forms a viscous pump that utilizes the viscous force developed by the difference in relative rotation between the bearing 124 and the rotating main shaft portion 122a. In general, since the viscous pump utilizes the viscous force to develop a powerful force of transportation, the freezer oil 103 can be assuredly pushed upwardly even at a low rotational speed of, for example, 20 Hz.

Hence, the freezer oil 103 reaching the third oil pump 161 is utilized to lubricate a sliding surface formed by an outer peripheral surface of the main shaft portion 122a and an inner peripheral surface of the bearing 124 and is also supplied

towards the eccentric shaft portion 122b.

On the other hand, the rotor 113 arranged to have its center of magnetization displaced downwardly from the center of magnetization of the stator 112 is, during the operation, shifted upwardly by the effect of a magnetic force of attraction as shown in Fig. 9 and, therefore, the first gap 131 will become almost zero over the entire circumference thereof. Accordingly, the freezer oil 103, which would flow outwardly from the first gap 131, can be drastically reduced and, therefore, the freezer oil 103 can be supplied to various sliding areas assuredly even at the low speed operation and, hence, the highly reliable hermetically sealed compressor could have been realized. As a result thereof, a stabilized supply of the oil can be achieved during a low speed operating condition, accompanied by a high reliability.

It is to be noted that although in describing each of the first to fourth embodiments, the reciprocating compressing mechanism has been shown and described, similar effects can nevertheless be obtained even with a scroll type compressing mechanism or with a rotary compressing mechanism.

Those functions and effects are universal regardless of the type of the coolant and that of the freezer oil.

Embodiment 5

Fig. 10 is a longitudinal sectional view of the hermetically sealed compressor according to a fifth preferred embodiment of the present invention, Fig. 11 is a cross-sectional view taken along the line A-A in Fig. 10, showing a rotor, and Fig. 12 is the sectional view of the rotor assumed to have an oil feed passage formed in the main shaft portion.

Referring to Figs. 10 and 11, a hermetically sealed vessel 201 is filled with a quantity of lubricant oil 202 and accommodates therein an electromotive element 203 and a compressing element 205 driven by the electromotive element 203. The compressing element 205 includes a shaft 210, having an eccentric shaft

portion 206 and a main shaft portion 207, and a main bearing 211 for supporting the main shaft portion 207.

A cylinder block 212 has a generally cylindrical compressing chamber 213 defined therein and has the main bearing 211 fixed thereto. A piston 214 is accommodated within the compressing chamber 213 in the cylinder block 212 for reciprocating movement therein and is drivingly connected with the eccentric shaft portion 206 through a connecting means 215.

A first oil pump 218 includes a hollow oil cone 219 fixed to a lower end of the main shaft portion 207 immersed in the lubricant oil 202, and an oil feed hole 220 perforated in a lower portion of the shaft 210, to thereby form a centrifugal pump.

A second oil pump 224 is disposed above the first oil pump 218 and is formed by a spiral groove 225, formed in an outer periphery of the main shaft portion 207, and an inner peripheral wall surface of a rotor 226 to thereby form an inertia pump. An upper portion of the first oil pump 218 and a lower portion of the second oil pump 224 are communicated with each other through a throughhole 227.

A third oil pump 228 is disposed above the second oil pump 224 and is formed by the spiral groove 225, formed in the outer periphery of the main shaft portion 207, and an inner peripheral surface of the main bearing 211, to thereby form a viscous pump.

The electromotive element 203 includes a stator 231 and the rotor 226 and is in the form of a bipolar permanent magnet synchronous motor of a self-starting type, in which the rotor 226 includes a rotor iron core 232 having a permanent magnet 234 built therein. Also, a protective end plate 235 for preventing the permanent magnet 234 from dropping out is fixed to the rotor iron core 232.

The bipolar permanent magnet electric motor is a permanent magnet synchronous motor of a self-starting type. In other words, it includes the rotor 226

of a structure, in which a plurality of conductor bars 241, provided in the rotor iron core 232, and a shortcircuiting ring 242 positioned in each of axially opposite ends of the rotor iron core 232 are formed integrally with each other by the use of an aluminum die casting technique to form a starter cage conductor, with a plurality of permanent magnets 234 embedded inside the starter cage conductor.

Each of the permanent magnets 234 is in the form of a flat plate and made of a ferromagnetic material such as containing boron, iron and neodymium which is a rare earth element. As shown in Fig. 11, the permanent magnets 234 of the same polarity are embedded axially in the rotor iron core 232 after having been so inserted and so arranged as to be held in angularly butted relation. Two permanent magnets 234 altogether define a single rotor magnetic pole and, hence, the entire permanent magnets 234 define two rotor magnetic poles. Also, in order to prevent magnetic fluxes from being shortcircuited between the neighboring permanent magnets 234, barriers 243 for avoiding the shortcircuiting of the magnets are formed, with an aluminum die cast filled in each of those barriers 243.

The coolant used in the compressor of the present invention is a hydrocarbon coolant, which is a natural coolant having a low global warming potential such as represented by R600a or R134a having a zero ozone depletion potential, and is used in combination with the lubricant oil having a high compatibility therewith.

The operation and effects of the hermetically sealed compressor of the structure described with reference to Figs. 10 to 12 will now be described.

When the rotor 226 of the electromotive element 203 drives the shaft 210 with the revolution of the eccentric shaft portion 206 transmitted to the piston 214 through the connecting means 213, the piston 214 undergoes a reciprocating movement within the compressing chamber 215. As a result, the coolant gas is, after having been sucked from a cooling system (not shown) into and then compressed within the compressing chamber 213, discharged again into the cooling

system.

The operation to supply the oil will be discussed hereinafter.

In the first oil pump 218, as the main shaft portion 207 rotates, the lubricant oil 202 is swirled within the oil cone 219 immersed in the lubricant oil 202, and the lubricant oil ascends along an inner peripheral surface of the oil cone 219 and that of the oil feed hole 220. The position of the throughhole 227 may be within a range of the main shaft portion 207, on which the rotor 226 is mounted, and is as low as possible, and, therefore, the capacity of the oil feed hole 220, which is a hollow in the main shaft portion 207, can be reduced so that the amount of magnetic fluxes as will be described later can advantageously be increased.

The lubricant oil 202 introduced from the first oil pump 218 into the second oil pump 224 through the throughhole 227 ascends within the spiral groove 225 in the second oil pump 224 by the effect of an inertia force developed by the inclination in the spiral groove 225 so as to act in an upward direction.

The lubricant oil 202 arriving at the third oil pump 228 ascends within the spiral groove 225 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 211 and the rotating main shaft portion 207. The lubricant oil 202 reaching the third oil pump 228 is utilized to lubricate a sliding surface formed by an outer peripheral surface of the main shaft portion 207 and an inner peripheral surface of the bearing 211 and is also supplied towards the eccentric shaft portion 206.

Accordingly, the capacity of the hollow in the main shaft portion 207 can be considerably reduced as compared with that in the prior art compressor and, therefore, the lubricant oil 202 can be assuredly supplied upwardly while the magnetic path within the main shaft portion 207 can be formed easily.

In the following description, the direction of flow of the magnetic fluxes of the permanent magnets will be conceptually described using arrow-headed lines in Figs. 11 and 12.

The flow of the magnetic fluxes appearing in the cross-section taken along the line A-A of the rotor iron core 232 is such that as shown in Fig. 11, the magnetic fluxes emanating from the upper two permanent magnets 234 as viewed in this figure pass through the center of the rotor iron core 232 and are absorbed in the lower two permanent magnets 234. On the other hand, the flow of the magnetic fluxes in the rotor iron core, which is assumed having a hollow oil feed passage defined in the main shaft portion such as in the prior art, is such that as shown in Fig. 12 the magnetic fluxes emanating from the upper two permanent magnets as viewed in the figure do not pass through the hollow oil feed passage, but pass radially outwardly of the hollow and, therefore, the magnetic fluxes at this portion tends to be insufficient.

However, in the embodiment now under discussion, since no hollow exists in the main shaft portion 204 as shown in Fig. 11, the magnetic path can be formed large in the main shaft portion 207 and, therefore, the amount of the magnetic fluxes inside the rotor iron core 232 increases, resulting in reduction of the loss.

Also, since the permanent magnets 234 are employed in the form of the rare earth magnets and the rare earth magnet in general can provide a strong magnetic force, not only can the electric motor be manufactured compact in size and light in weight, but the hermetically sealed compressor can also be manufactured compact in size.

Accordingly, even where the bipolar permanent magnet electric motor is employed in the hermetically sealed compressor having the compressing element 205 disposed in an upper region, the amount of the magnetic fluxes emanating from the permanent magnets can be increased, allowing the hermetically sealed compressor to be compact in size and lightweight and also to have a high efficiency.

Embodiment 6

Fig. 13 is a longitudinal sectional view of the hermetically sealed

compressor according to a sixth preferred embodiment of the present invention. Fig. 14 is a cross-sectional view of the rotor taken along the line B-B in Fig. 13.

Referring to Figs. 13 and 14, a hermetically sealed vessel 301 is filled with a quantity of lubricant oil 302 and accommodates therein an electromotive element 303 and a compressing element 305 driven by the electromotive element 303. The compressing element 305 includes a shaft 309 having an eccentric shaft portion 306, a main shaft portion 307 and an auxiliary shaft portion 308, in which the auxiliary shaft portion lies coaxially with the main shaft portion 307 with the eccentric shaft portion 306 intervening between it and the main shaft portion 307. The main shaft portion 307 is supported by a main bearing 310 and the auxiliary shaft portion 308 is supported by an auxiliary bearing 311.

The main bearing 310 is of a structure which does not intersect the imaginary plane containing one end of the rotor iron core 312 adjacent the compressing element 305 and lying generally perpendicular to the longitudinal axis of the main shaft portion 307. In other words, the main bearing 310 has an axial length which is somewhat reduced, so that it will not protrude inwardly of the rotor iron core 312, and no hollow is defined in that end of the rotor iron core 312 adjacent the compressing element 305.

A cylinder block 313 has the auxiliary bearing 311 and a generally cylindrical compressing chamber 314 defined therein and has the main bearing 310 fixed thereto. A piston 315 is accommodated within the compressing chamber 314 in the cylinder block 313 for reciprocating movement therein and is drivingly connected with the eccentric shaft portion 306 through a connecting means 316.

A first oil pump 318 includes a hollow oil cone 319 fixed to a lower end of the main shaft portion 307 immersed in the lubricant oil 302, and an oil feed hole 320 perforated in a lower portion of the shaft 309, to thereby form a centrifugal pump.

A second oil pump 324 is disposed above the first oil pump 318 and is

formed by a spiral groove 325, formed in an outer periphery of the main shaft portion 307, and an inner peripheral wall surface of a rotor 326, to thereby form an inertia pump. An upper portion of the first oil pump 318 and a lower portion of the second oil pump 324 are communicated with each other through a throughhole 327.

A third oil pump 328 is disposed above the second oil pump 324 and is formed by the spiral groove 325, formed in the outer periphery of the main shaft portion 307, and an inner peripheral surface of the main bearing 310, to thereby form a viscous pump.

The electromotive element 303 includes a stator 331 and the rotor 326 and is in the form of a bipolar permanent magnet synchronous motor of a self-starting type, in which the rotor 326 includes a rotor iron core 312 having a permanent magnet 334 built therein. Also, a protective end plate 335 for preventing the permanent magnet 334 from dropping out is fixed to the rotor iron core 312.

The bipolar permanent magnet electric motor is a permanent magnet synchronous motor of a self-starting type. In other words, it includes the rotor 326 of a structure, in which a plurality of conductor bars 341, provided in the rotor iron core 312, and a shortcircuiting ring 342 positioned in each of axially opposite ends of the rotor iron core 312 are formed integrally with each other by the use of an aluminum die casting technique to form a starter cage conductor, with a plurality of permanent magnets 334 embedded inside the starter cage conductor.

Each of the permanent magnets 334 is in the form of a flat plate and made of a ferromagnetic material such as containing boron, iron and neodymium which is a rare earth element. As shown in Fig. 14, the permanent magnets 334 of the same polarity are embedded axially in the rotor iron core 312 after having been so inserted and so arranged as to be held in angularly butted relation. Two permanent magnets 334 altogether define a single rotor magnetic pole and, hence, the entire permanent magnets 334 define two rotor magnetic poles. Also, in order

to prevent magnetic fluxes from being shortcircuited between the neighboring permanent magnets 334, barriers 343 for avoiding the shortcircuiting of the magnets are formed, with an aluminum die cast filled in each of those barriers 343.

The coolant used in the compressor of the present invention is a hydrocarbon coolant, which is a natural coolant having a low global warming potential such as represented by R600a or R134a having a zero ozone depletion potential, and is used in combination with the lubricant oil having a high compatibility therewith.

The operation of the hermetically sealed compressor of the structure described with reference to Figs. 13 and 14 will now be described.

When the rotor 326 of the electromotive element 303 drives the shaft 309 with the revolution of the eccentric shaft portion 306 transmitted to the piston 315 through the connecting means 316, the piston 315 undergoes a reciprocating movement within the compressing chamber 314. As a result, the coolant gas is, after having been sucked from a cooling system (not shown) into and then compressed within the compressing chamber 314, discharged again into the cooling system.

The operation to supply the oil will be discussed hereinafter.

In the first oil pump 318, as the main shaft portion 307 rotates, the lubricant oil 302 is swirled within the oil cone 319 immersed in the lubricant oil 302, and the lubricant oil ascends along an inner peripheral surface of the oil cone 319 and that of the oil feed hole 320. The position of the throughhole 327 may be within a range of the main shaft portion 307, on which the rotor 326 is mounted, and is as low as possible, and, therefore, the capacity of the oil feed hole 320, which is a hollow in the main shaft portion 307, can be reduced so that the amount of magnetic fluxes as will be described later can advantageously be increased.

The lubricant oil 302 introduced from the first oil pump 318 into the second oil pump 324 through the throughhole 327 ascends within the spiral groove

325 in the second oil pump 324 by the effect of an inertia force developed by the inclination in the spiral groove 325 so as to act in an upward direction.

The lubricant oil 302 arriving at the third oil pump 328 ascends within the spiral groove 325 by the effect of a viscous force developed by the difference in relative rotation between the fixed bearing 310 and the rotating main shaft portion 307. The lubricant oil 302 reaching the third oil pump 328 is utilized to lubricate a sliding surface formed by an outer peripheral surface of the main shaft portion 307 and an inner peripheral surface of the bearing 310 and is also supplied towards the eccentric shaft portion 306 and the auxiliary shaft portion 308.

Accordingly, the capacity of the hollow in the main shaft portion 307 can be considerably reduced as compared with that in the prior art compressor and, therefore, the lubricant oil 302 can be assuredly supplied upwardly while the magnetic path within the main shaft portion 307 can be formed easily.

In the following description, the direction of flow of the magnetic fluxes of the permanent magnets will be conceptually described using arrow-headed lines in Fig. 14. The flow of the magnetic fluxes appearing in the cross-section taken along the line B-B of the rotor iron core 312 is such that as shown in Fig. 14, the magnetic fluxes emanating from the upper two permanent magnets 334 as viewed in this figure pass through the center of the rotor iron core 312 and are absorbed in the lower two permanent magnets 334.

On the other hand, the flow of the magnetic fluxes in the rotor iron core, which is assumed having a hollow oil feed passage defined in the main shaft portion such as in the prior art, is such that as shown in Fig. 12 the magnetic fluxes emanating from the upper two permanent magnets as viewed in the figure do not pass through the hollow oil feed passage, but pass radially outwardly of the hollow and, therefore, the magnetic fluxes at this portion tends to be insufficient. However, in the embodiment now under discussion, since no hollow exists in the main shaft portion 307 as shown in Fig. 14, the magnetic path can be formed large in the main

shaft portion 307 and, therefore, the amount of the magnetic fluxes inside the rotor iron core 312 increases, resulting in reduction of the loss.

Also, since the main bearing 310 is of a structure which does not intersect the imaginary plane containing one end of the rotor iron core 312 adjacent the compressing element 305 and lying generally perpendicular to the longitudinal axis of the main shaft portion 307, no hollow bore is defined in the main bearing, which has hitherto been required in the prior art so that the main bearing 310 can be inserted into the rotor iron core 312. As a result thereof, a possible narrowing of the magnetic path, which would be brought about by the presence of the hollow bore, can be eliminated and, therefore, the amount of the magnetic fluxes inside the rotor iron core 312 increases further, accompanied by increase of the efficiency.

In addition, the shaft 309 is provided, which has the eccentric shaft portion 306, the main shaft portion 307 and the auxiliary shaft portion 308. The auxiliary shaft portion 308 is held in coaxial relation with the main shaft portion 307 with the eccentric shaft portion 306 intervening between it and the main shaft portion 306, the main shaft portion 307 is supported by the main bearing 310, and the auxiliary shaft portion 308 is supported by the auxiliary bearing 311.

Accordingly, without insertion of the main bearing 310, of which length is reduced, into the rotor iron core 312 in order to fundamentally regulate the inclination of the shaft 309, the inclination of the shaft 309 is extremely minimal and there is no possibility that the shaft 309 and the main bearing 310 or the auxiliary bearing 311 become complicated and, therefore, the noise can be lowered with the reliability and the efficiency increased.

Yet, since the permanent magnets 334 are employed in the form of the rare earth magnets and the rare earth magnet in general can provide a strong magnetic force, not only can the electric motor be manufactured compact in size and light in weight, but the hermetically sealed compressor can also be manufactured compact in size.

Accordingly, even where the bipolar permanent magnet electric motor is employed in the hermetically sealed compressor having the compressing element 305 disposed in an upper region, the amount of the magnetic fluxes emanating from the permanent magnets can be increased, allowing the hermetically sealed compressor to be compact in size, lightweight, high in efficiency, less noisy, and highly reliable.

Industrial Applicability

The hermetically sealed compressor of the present invention is effective to accomplish a stabilized oil supply during the low speed operating condition and is useful as a hermetically sealed compressor for use with a freezer/refrigerator such as used in a humidifier, showcase, automatic vending machine, air conditioner, since the amount of the magnetic fluxes inside the rotor iron core increases with the loss reduced and since the efficiency can be increased with a compact and light weight structure.